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RELIABILITY OF EXISTING METHODS FOR CALCULATING THE HEAT TRANSFER BETWEEN THE GAS AND A COOLED TURBINE BLADE

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RELIABILITY OF EXISTING METHODS FOR CALCULATING THE HEAT TRANSFER BETWEEN THE GAS AND A COOLED TURBINE BLADE

Ye. P. Dyban and V. G. Glushchenko

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Many of the relationships recommended in the literature for determining the average and local heat coefficients of transfer from a gas to a profile of turbine blades have been obtained in experiments with a low value of the temperature factor, the Mach number and frequently with an inverse direction of the thermal flow. Therefore, it is necessary to verify the recommended computational methods based on the results of tests of cooled blades under conditions which approximate real conditions to the maximum extent.

This article uses the results of experiments on fixed groups of turbine blades immersed in a hot bath and cooled by air, in order to solve this problem. In the experiments, the ratio between the temperatures of the gas and the blade walls was 1.8 - 1.32. The Mach number changed from 0.32 to 1.092, and the Reynolds number (determined along the chord and the flow parameters at the output) changed from $2.15 \cdot 10^5$ to $8.98 \cdot 10^5$.

The experimental equipment, the test method, the geometric characteristics of the profiles studied and the method of determining the heat transfer coefficients were described in [10].

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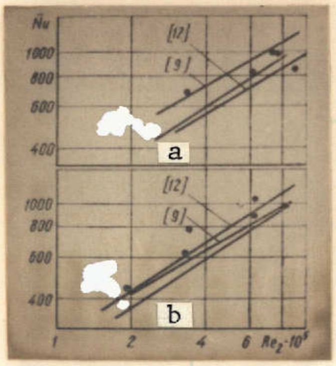

Figure 1. Average heat exchange between a gas and the profile of the turbine rings studied

Figure 1 gives the values of the average Nusselt numbers along the blade perimeters, which were obtained by integration of the curves obtained during the experiments for the local heat transfer coefficients. The blade chord was used as a characteristic linear dimension for both rings when determining the Nusselt and Reynolds numbers. The Reynolds number was determined from the flow parameters at the ring output, and the heat conductivity and dynamic viscosity coefficients were determined from the total gas temperature. The dependences for calculating the average heat exchange are also plotted in Figure 1 [8, 9, 12].

It may be seen from Figure 1 that the experimental points for ring I (Figure 1, a) lie below the generalized dependence [9] by an average of 15% and above the generalized dependence [12] by an average of 16%. The experimental values of the average heat transfer coefficients in ring II (Figure 1, b) lie above the calculated values given in [9] by 14% and above the values determined in [12] by 5%.

The dependences given in [9, 12] most completely take into account the geometric characteristics of the rings, and for the blades studied give different dependences $\overline{Nu} = f(Re_2)$. The dependence in [8] which takes into account only the sum of the input and output angles of the blade gives one dependence $\overline{Nu} = f(Re_2)$ for both of the rings studied. Due to this, the difference between experimental values and the values calculated in [8] is 25% in ring II and 12% in ring I.

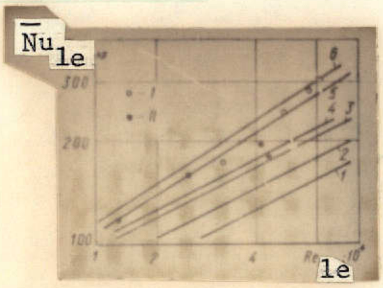

Figure 2. Heat exchange between gas and leading edge of turbine blade:

1- calculation for blades I and II, according to [12]; 2- calculation for blades I and II according to [8]; 3- calculation for blade II according to [14] when Ti = 0; 4- calculation for blade I according to [14] when Ti = 0; 5- calculation for blade I according to [13,14] when Ti = 2%; 6 - calculation for blade II according to [13, 14] when Ti = 2%

The literature gives two basic methods for determining the local heat transfer coefficients: "the method of several (3 or 4) regions" [8, 12], and the method based on calculating the boundary layer [5]. Although the method of several regions frequently pertains to the methods of calculating the local heat transfer coefficients, it can be used to calculate only the heat transfer coefficients for individual profile regions (input and output edges, middle region of the profile perimeter). /30

In [8] the heat exchange at the leading edge is calculated according to a criterial equation for a cylinder in a transverse flow, on the middle region of the profile — according to a criterial equation which describes the average heat exchange on the blade perimeter, and at the trailing edge — according to the equation for a plane plate washed by a turbulent boundary layer.

The study [12] uses the same kind of criterial equation as was used in [8] to calculate the heat exchange at the leading edge, but the numerical coefficient was determined in experiments.

The heat transfer of the middle region of the blade in [12] was determined by means of a criterial equation for the average heat exchange with subsequent correction (individually for concave and convex surfaces) as a function of the ring degree of reactivity.

For the surface of the trailing edge, with the radius r_t at a distance to 0.1 b_0 (b_0- profile chord), in [12] the heat exchange is calculated from the criterial equation obtained in experiments.

There are no data in the literature on verifying the methods proposed in [8, 12] for performing calculations based on the results of studies of real blades.

The method of determining the local heat transfer coefficients in [5] is the most valid from the physical point of view, since it takes into account the influence of the largest number of parameters upon the heat exchange. However, the criterial dependences used in it to determine the absolute values of the heat transfer coefficients, and also the coordinates of the transition region [6], were obtained for a plate under idealized conditions. The possibility of using them for conditions of real turbines may only be established experimentally.

It is stated in [1, 2], on the basis of experiments on model blades with inverse direction of the thermal flow, small Mach numbers ($M<0.72$), and temperature heads, that the results of calculations agree satisfactorily with experiments when the coordinates of the transition region are determined directly from experimental data. It is indicated in [11] that in this case the results of calculations in [5] deviate from its experimental data by 20-40%. Figure 2 gives the values of $\overline{Nu}_{le}$ for both blades studied. The Reynolds numbers were determined in accordance with the recommendations in [8, 12] on the flow parameters at the ring input and the diameter of the blade leading edge. The same figure plots the criterial dependences recommended in [8, 12] for calculating the heat exchange at the leading edge. On the average, the experimental points are 40% above those given in [8], and 47% above those given in [12].

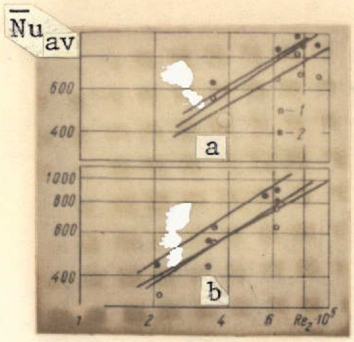

Figure 3. Heat exchange on the middle section of the profile:

1- convex surface; 2- concave surface (according to [12]).

It must be noted that the criterial relationships used in these studies do not take into account the influence of the turbulence Tu of the incoming flow and the heat exchange. The use of recommendations given in [13] in the calculations makes it possible to decrease the deviation of the experimental data from the calculated values and when Tu = 2% (the degree of turbulence proposed in our experiments) the experimental data differ from the calculated values given in [13, 14] by 8-10% for both blades (see Figure 2). / 31

Figure 3 gives the values of $\overline{Nu}_{av}$ for the middle region of the blades studied (determined by averaging over the local heat transfer coefficients). In both blades, the heat transfer coefficients (the Nusselt numbers) for convex and concave surfaces differ substantially. Thus, for ring I (Figure 3, a) the heat transfer coefficients for a concave surface (solid dots) are on the average 25% above those for a convex surface (light dots), and for ring II (see Figure 3, b) — 30% higher. This same figure plots the dependence recommended in [8] for calculating the heat exchange on the profile. As was already indicated, the methods of several regions used it for determining the heat transfer coefficients on the middle region of the profile.

It can be seen in Figure 3 that the divergence between the calculated [8] and experimental data for ring II on convex and concave surfaces is 28 and 7% and for ring II — 15 and 13%, respectively. With allowance for the recommendations in [12] on the correction of the local heat transfer coefficients, the divergence for convex and concave surfaces for blade I is 4 - 5% and for blade II — 12 - 6%, respectively.

One of the great disadvantages of calculating the local heat transfer coefficients using the method of several regions is the lack of precise recommendations on defining the concept of the "trailing edge". Sometimes the trailing edge is assumed to be part of the profile perimeter (adjacent to the trailing edge) with a length of 1/3 of the chord. In other cases, the trailing edge designates the region of the blades lying behind the section which coincides with the "throat" of the ring, in [12] — the region of the latter which is approximated by a circular cylinder together with the region of the perimeter which has a length up to 0.1 of the chord.

In this study, the local heat transfer coefficients at the trailing edge were determined for regions in which they increase monotonically (from S/L = 0 to S/L = 0.16- 0.2 for a convex surface and from S/L = 0.8 - 0.88 to S/L = 1.0 for a concave surface). Figure 4 gives the values of $\overline{Nu}_{te}$ which are averaged in this way, as well as the dependence given in [8] for calculating the heat exchange on the trailing edge. For blades with discharge of cooling air directly over the trailing edge, it is not possible to make a comparison with the dependences given in [2, 12].

For $Re_2 = 2 \cdot 10^5$ numbers, the experimental values of 50 lie 50% above the calculated values. The great difference between the calculated and experimental values of α_t may be explained primarily by the fact that the dependence given in [8] characterizes the heat exchange on a plate with a turbulent boundary layer, whereas the flow at the trailing edge may differ from a turbulent flow.

The data obtained in this investigation for the average heat transfer coefficients in the region of the trailing edge may be described by the following empirical relationships:

For a convex surface

$$Nu_t\ M_{2_{AD}} = 0.5 = 6.15\ Re_2^{0.4} \left(\frac{T_B}{T^*_{th}}\right)^{-0.5}, \qquad (1)$$

For a concave surface

$$Nu_t\ M_{2_{AD}} = 0.5 = 0.535\ Re_2^{0.6} \left(\frac{T_B}{T^*_{th}}\right)^{-0.5}, \qquad (2)$$

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In these relationships, the chord of the blade is the geometric dimension, the Reynolds number is determined from the parameters at the output of the ring, and the thermophysical characteristics are determined from the gas temperature. The number of the exponent in the case of the Reynolds number provides a basis for assuming that there is detached flow both from the convex and from the concave sides of the trailing edge of the blades. The weaker dependence of the heat transfer coefficients on the Reynolds number on the convex surface may be explained by the diffused flow of this region of the interblade channel, as compared with a concave surface where convergent flow occurs.

It follows from a comparison of the heat transfer coefficients on the trailing edge that when there is detached flow the calculation of the heat transfer coefficients, according to the recommendations given in [8] may lead to great errors, and it is not permissible to determine the heat transfer coefficients for the trailing edge of the blade as a whole, without separating the convex and concave surfaces. Since the previous history of the boundary layers on these surfaces differs greatly, this may lead (as was the case in our experiments) to a great difference

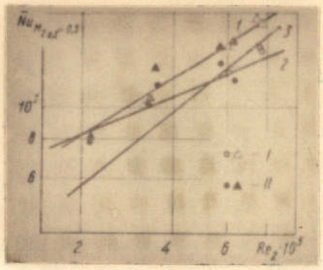

Figure 4. Heat exchange between gas and trailing edge of turbine blade:

1- concave surface [Equation (2)]; 2- convex surface [Equation (1)]; 3- calculation for blades I and II according to [8]

in the values of the heat transfer coefficients.

Figures 5 and 6 compare the heat transfer coefficients obtained* for blades I and II with allowance for the method given in [5]. The curves of the relative velocities, which are necessary for the calculations, were determined from the /33 experimental data (drainage of the blades). The influence of the Mach number upon a change in the flow around the rings was taken into account.

The value of α_1 for a convex surface was calculated twice. In the first variation, the coordinate of the beginning of the transition region was determined from a dependence given in [3], and the coordinate of the end of the transition region was determined from recommendations in [7]. In this method of determining the coordinates of the transition region on the convex surfaces of both blades, there are no regions with a turbulent boundary layer. As a result of the fact that the calculated coordinates of the transition region do not coincide with the real coordinates, there is maximum divergence between the calculated and experimental values of the heat transfer coefficient. For blade I the divergence is 300%, and for blade II — 380%.

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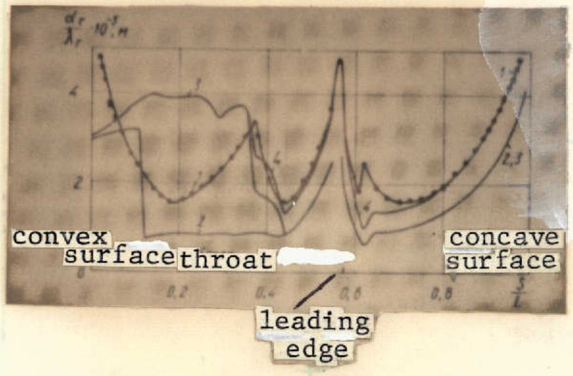
* Figures 5 and 6 compare the calculated and experimental data for one Reynolds number, and in the remaining regimes studied the calculated and experimental curves have approximately the same distribution.



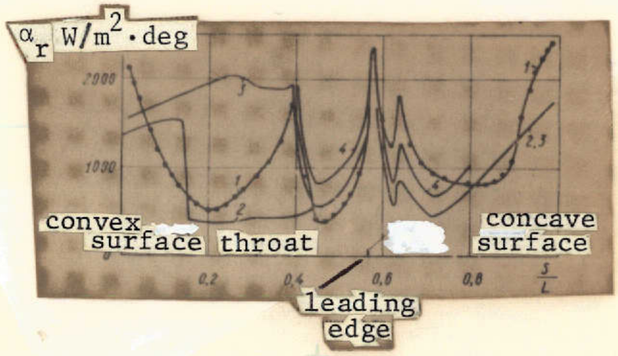


Figure 5. Distribution of local heat transfer coefficients along the perimeter of ring I,

1- experimental values; 2- calculation according to [5] with determination of the transition region coordinates according to [3, 7]; 3- calculation according to [5] when the transition region coordinates are given according to experimental data; 4- calculation similar to 3, with allowance for recommendations in [11]

Figure 6. Distribution of local heat transfer coefficients along the perimeter of ring II (notation is the same as in Figure 5).

In the second variation, the coordinate of the transition region was calculated directly from the local heat transfer coefficients obtained when solving the inverse problems of heat conductivity* (curve 3). In this case, in the regions of the concave surface of the blades with the relative coordinate of 0.7- 1.0, the divergence between experimental and calculated values of the heat transfer coefficients was 20 - 100%. With the

* The coordinates of the beginning of the transition region [6], when the turbulence values given above were used, differed for both blades from the experimental values by ±7.0%.

exception of the region of the blade II with the relative coordinates of 0.8 - 0.9, the experimental values for the heat transfer coefficients were higher than the calculated values. In the remaining regions of the concave surface, the divergence between experimental and calculated values exceeded 100%.

It may be seen from a comparison of curves 1 and 3 that in the regions of the convex surface with laminar and transition boundary layers, on the average the difference is ± 20%.

The greatest difference between the calculated and experimental data occurs in regions with a turbulent boundary layer (S/L = 0.19 - 0.37 for blade I and S/L = 0.2 - 0.4 for blade II. Not only a quantitative but also a qualitative change is characteristic in the curves for the heat transfer coefficients in these regions.

The following may be noted when analyzing the reasons for the divergence between the calculated and the experimental values of the local heat transfer coefficients.

The influence of the turbulence of the external flow upon the heat exchange intensity was not taken into account in [5]. As was shown in [11], this may be observed in laminar and transition boundary layers.

The heat transfer coefficients for regions of a blade lying below the transition region were calculated in [5] using relations obtained for a plate in a turbulent flow (with allowance for the velocity curve). The presence of detached flow in this region was not considered in [5].

If it is assumed that, in agreement with [11], for regions with laminar, transition and turbulent flow in the boundary layer the value of $\varepsilon_{Ti} = Nu_{Ti}/Nu_{Ti=0} = 1.4$, then — as may be seen from Figures 5 and 6 (curve 4) — the difference (in percents) between the experimental and calculated (with allowance for ε_{Ti}) values of Nu on a concave surface of both blades (up to a point with the coordinate $S/L = 0.8$) decreases by almost a factor of two. The calculated and experimental curves thus practically coincide for a convex surface of blade I in regions with laminar and transition flows.

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16. Abstract

Local and mean gas/blade heat transfer coefficients calculated by several available methods are compared with experimental data obtained in the solution of inverse steady state heat conduction problems for two rings of cooled stator blades. It is shown that the observed discrepancy between the calculated and experimental coefficients at the leading edges and at blade sections with laminar, transition and turbulent boundary layers can be greatly reduced by allowing for the degree of external flow turbulence. Two empirical relations, one for convex and one for concave surfaces are proposed for calculating heat transfer at blade trailing edges.

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